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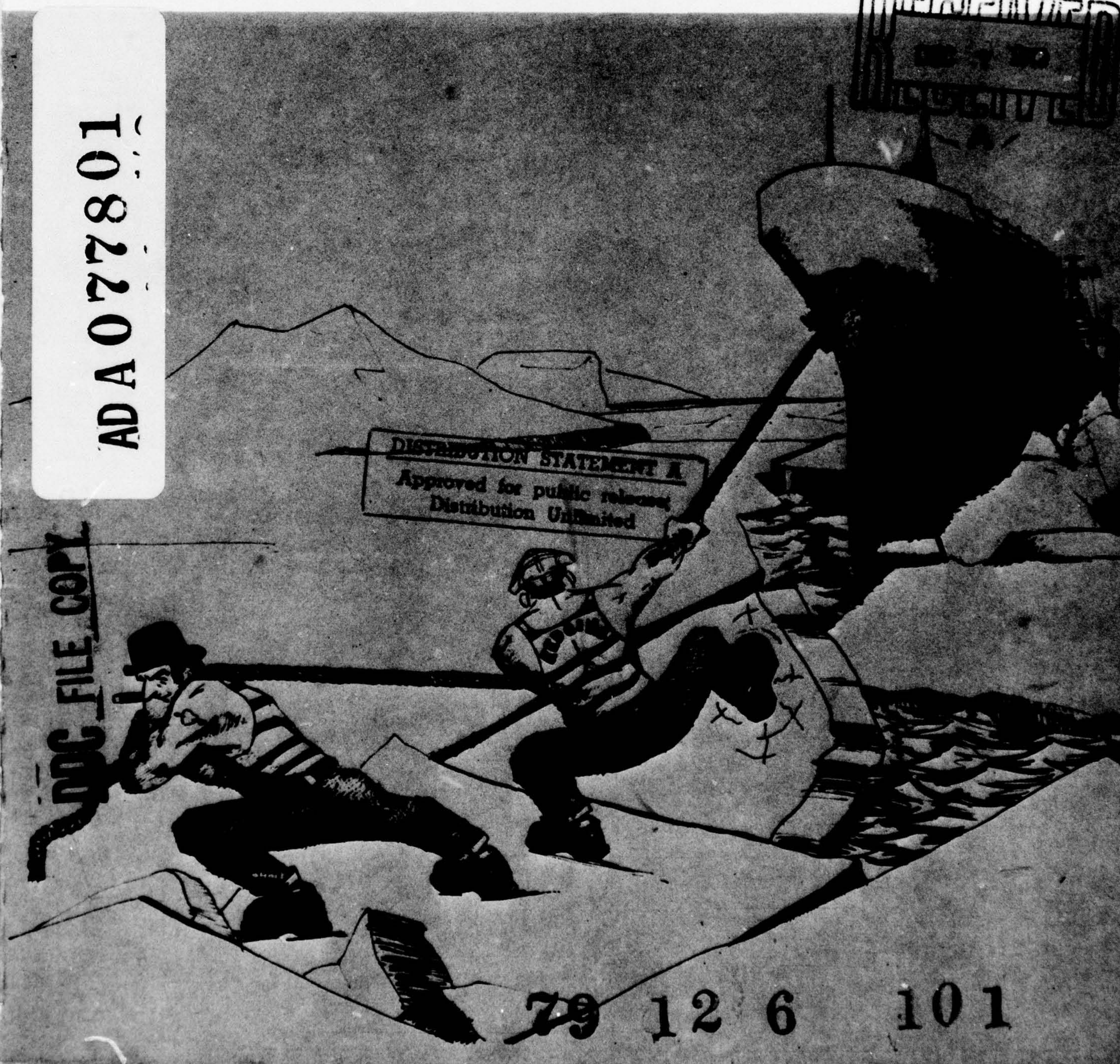

CRREL Report 79-21

Towing ships through ice-clogged channels by warping and kedging

Malcolm Mellor

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The report studies the question of whether Great Lakes freighters could move effectively through ice-clogged channels with the aid of tows provided by warping or kedging systems. Ten operational concepts are outlined, and their advantages and disadvantages are noted. The crushing resistance of floating brash ice is then analyzed. The neutral, active and passive states of stress for laterally confined brash ice are considered, and the resistance to horizontal thrusting by a smooth vertical wall is calculated for cohesionless brash ice, and for ice in which there is finite cohesion between the ice fragments. The thickening of the ice cover in the vicinity of a "pusher," and the formation of pressure ridges, are analyzed in order to estimate the amount of pile-up that can occur against a ship hull. The analysis then moves on to

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20. Abstract (cont'd).

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PREFACE

This report was prepared by Dr. Malcolm Mellor, Physical Scientist, Experimental Engineering Division, U.S. Army Cold Regions Research and Engineering Laboratory. The work was carried out under U.S. Coast Guard contract no. A70099-8334-2013. The report was reviewed by Dr. George Vance and Dr. Devinder Sodhi of CRREL.

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TOWING SHIPS THROUGH ICE-CLOGGED CHANNELS BY WARPING AND KEDGING

Malcolm Mellor

ABSTRACT

The report studies the question of whether Great Lakes freighters could move effectively through ice-clogged channels with the aid of tows provided by warping or kedging systems. Ten operational concepts are outlined, and their advantages and disadvantages are noted. The crushing resistance of floating brash ice is then analyzed. The neutral, active and passive states of stress for laterally confined brash ice are considered, and the resistance to horizontal thrusting by a smooth vertical wall is calculated for cohesionless brash ice, and for ice in which there is finite cohesion between the ice fragments. The thickening of the ice cover in the vicinity of a "pusher," and the formation of pressure ridges, are analyzed in order to estimate the amount of pile-up that can occur against a ship hull. The analysis then moves on to consideration of ship resistance by brash ice, taking into account crushing resistance at the bow, tangential friction at the bow, and hull friction aft of the bow section. Comparisons are made between thrust from the ship's screws and the calculated ice resistance. The next section of the report estimates the force requirements for a warping or kedging system in terms of thrust augmentation for existing vessels. Tow cable requirements are given, and estimates are made for cable anchors and for anchorage of underwater structures. The force and power requirements for winches and windlasses are given, the practical problems involved in the pickup or transfer of cables are mentioned, and the report concludes with a brief appraisal. The conclusion is that a simple warping tug system is appropriate for a full scale experiment, a chain ferry with auxiliary barge seems attractive for an operational system, and a chain ferry plow may be an efficient way to clear ice from channels.

INTRODUCTION

The aim of this study is to determine whether it might be useful and feasible to tow Great Lakes freighters through heavy concentrations of brash ice by using warping or kedging methods.

The first part of the report puts forward for consideration a variety of operational schemes that might conceivably be adopted, and lists some advantages and disadvantages. It should be stressed that these ideas have been outlined without benefit of first hand experience of the field situation

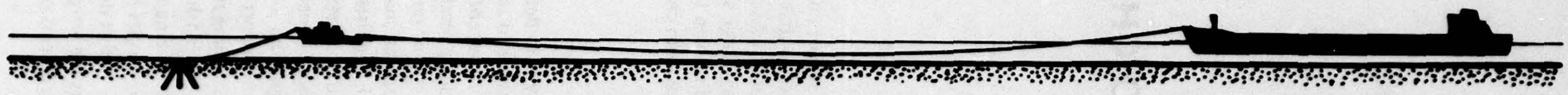
The second part of the report is an original analytical study dealing with the crushing resistance of brash ice and with ship resistance in brash ice. This work was prompted by lack of convincing theoretical material in the literature, and the results need checking.

The third part of the report deals with the force and power requirements for warping and kedging systems, and with some of the problems of design, procurement, and operation.

OPERATIONAL CONCEPTS

(N.B. In all schemes, the transient vessel is expected to use its own screw propulsion to reduce towing force.)

A. Warping Tug System



Procedure:

A powerful tug with a traction winch (e.g. an anchor-handling tug) is stationed for the winter season in the problem area. On arrival of a transient vessel, the tug passes its main tow line to the vessel, then moves through the ice-clogged channel up to the anchor point while paying out the towing cable. The tug hooks on to the mooring buoy and winches the vessel through the channel.

Advantages:

Simple to deploy.

Little development effort and small capital investment (assuming tug could be chartered).

Could be put into operation immediately and could be abandoned with little loss if unsatisfactory.

No modifications required by transient vessels; no special crew skills.

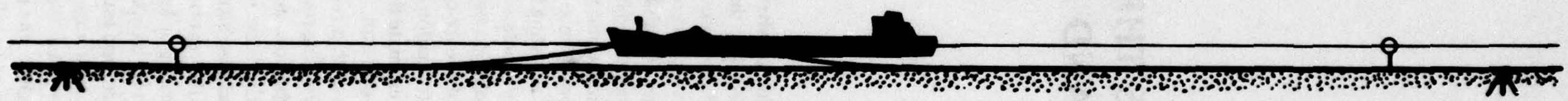
Disadvantages:

Relatively high operating cost (equipment, labor, fuel).

Tug itself might have difficulties in heavy ice.

Would probably be restricted to a few critical areas because of cost.

B. Chain Ferry System



Procedure:

A chain or flexible cable is laid slack on the bed of the channel between two end anchors. On arrival of a transient vessel, the cable is picked up at the marker buoy, and it is engaged to a special outrigged capstan or windlass on the vessel. The vessel then hauls itself along the cable until the other anchor point is reached.

Advantages:

Cheap and simple for the waterway authority, with virtually no operating cost.

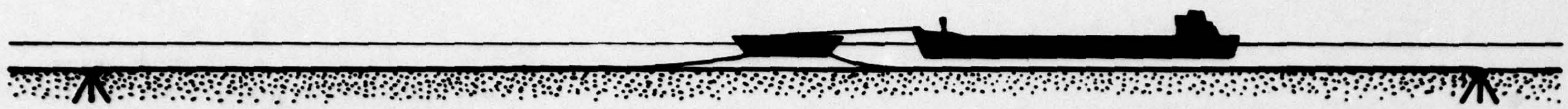
Could be installed in many places without incurring major costs.

Disadvantages:

All vessels would have to be fitted with suitable lifting gear and traction capstans.

Some crew training required.

C. Chain Ferry with Auxiliary Barge



Procedure:

A chain or flexible cable is laid slack on the bed of the channel between end anchors. A barge or pontoon fitted with a special capstan or windlass is attached to the chain or cable permanently (i.e. during the ice season). On arrival of a transient vessel, a short tow cable is passed between that vessel and the pontoon. The pontoon hauls itself along the cable, pulling the ship in its wake.

Advantages:

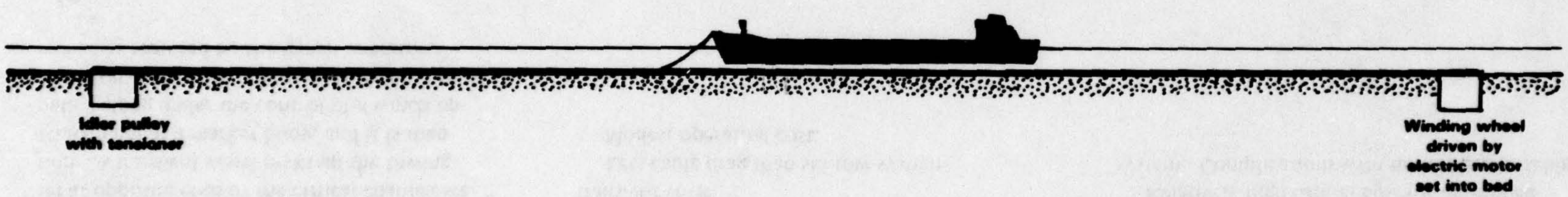
Simple to deploy and operate.

No underwater machinery.

Disadvantages:

Some operating problems might have to be defined and solved by prototype system.

D. Ski Tow System



Procedure:

A continuous loop of tensioned cable is laid flat on the bed of the channel. At one end it passes around an idler pulley which has a cable tensioning device. At the other end it passes around a drive wheel, which is powered by an electric (or possibly hydraulic) motor. The cable is driven on demand by an operator on shore.

On arrival, the transient vessel picks up the towing attachment at the marker buoy, and is then drawn along to the opposite end of the loop. One or more towing attachments are fixed to the main cable by special offset grippers.

Advantages:

No special equipment or special skills required on transient vessel.

Modest operating cost.

Disadvantages:

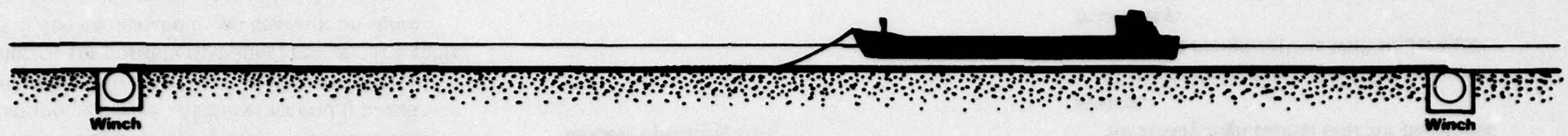
Appreciable time and money needed for initial development.

Relatively high capital cost for developed system.

Distance limited by cable friction on bed.

Inherent complications with underwater machinery.

E. Dual Winch Warping System



Procedure:

A cable is laid on the bed between two winches set at opposite ends of the critical channel section. A transient vessel picks up the towing attachment at a marker buoy, and it is then pulled along under the control of a winch operator on shore. The towing attachment can be moved as required by the winch operator.

Advantages:

No special equipment or skills required on transient vessel.

Less cable drag than ski tow system.

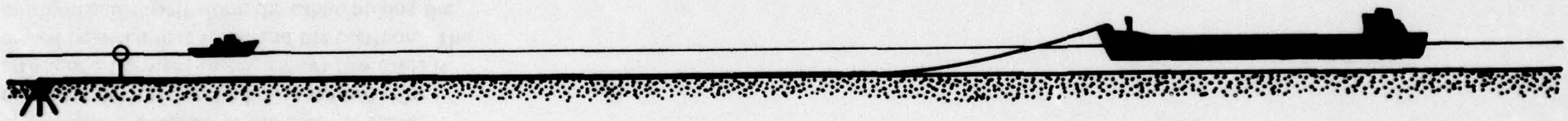
Modest operating cost.

Disadvantages:

Time and money needed for initial development.

Relatively high capital cost for developed system. Complications with underwater machinery.

F. Simple Kedging



Procedure:

A winch cable on an arriving transient vessel is passed to a local tug, which carries the cable forward and attaches it to a permanent anchor. The vessel then pulls itself forward using its own traction winch.

Advantages:

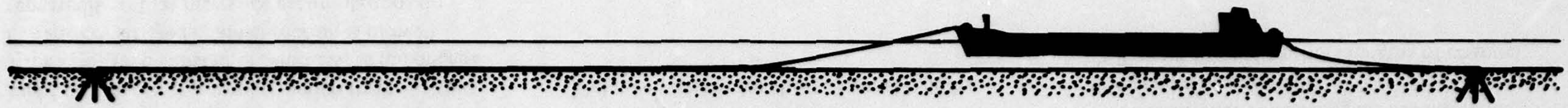
Relatively simple for waterway authority.

Disadvantages:

All vessels would have to be fitted with traction winches.

Appreciable cost for both users and operators.

G. Trailing-Line System



Procedure:

Separate cables are attached to permanent anchors located at opposite ends of the critical channel section. Each cable is attached permanently to a storage drum, which can be fitted to standardized traction winches on transient vessels. The cable drum might also serve as a buoy. On arrival, a transient vessel picks up two cable drums, fitting the "active" one to its traction winch and the "full" one to its payout reel. It kedges itself forward and drops both drums and their anchors at the opposite end of the channel, having clamped the cables to the drums.

Advantages:

Relatively cheap for the waterway authority.

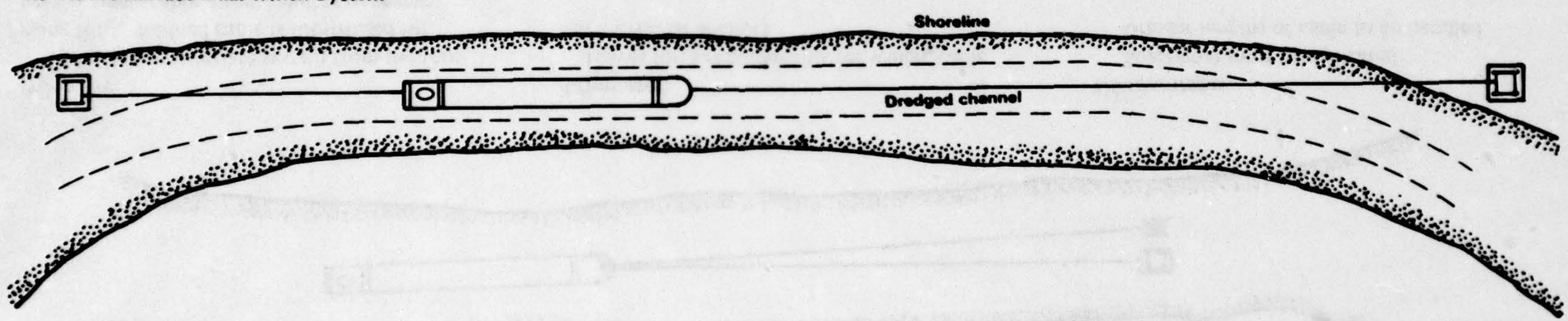
Disadvantages:

Would require strict alternation of traffic at a 1:1 ratio.

Complicated and awkward to use.

Expensive equipment required on vessels.

H. Above-Surface Dual Winch System



Procedure:

Where channel conditions are favorable, winch stations are built either on shore or in shallow water out of the ship lane. In other respects, the procedure is similar to that described for the underwater dual winch system (E).

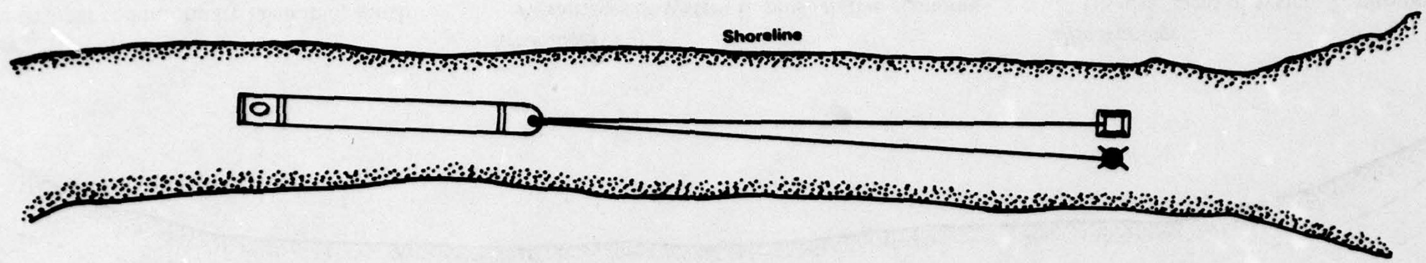
Advantages:

Advantages of system E, plus relative economy of above-surface construction and operation for winch stations.

Disadvantages:

Disadvantages of system E, plus restriction to favorable channel situations.

I. Pulley Systems



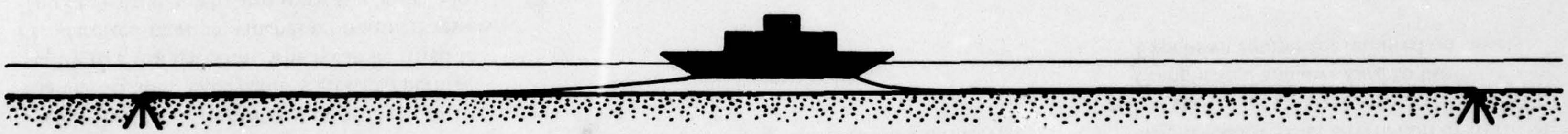
Procedure:
Using any appropriate system from the foregoing list, a doubled cable is substituted for heavier single cable, using a pulley or sheave.

Advantages:
Halves force requirements for winch, cable and individual anchors.

Disadvantages:
Somewhat more complicated.
Greater lengths of cable to be handled.

NOTE: For given transit speed winch speed doubles and power demand remains unaltered.

J. Chain Ferry Plow



Procedure:
A wide floating V-plow, say 200 ft wide, is build on a barge or pontoon, which itself is fitted with a powered capstan or windlass, as in concept C. The constricted channel section is plowed repeatedly so that heavy ice accumulations do not develop.

Advantages:
No delay for transient vessels.
Relatively cheap for short channel lengths.

Disadvantages:
Uncertainties concerning ice accumulation along edges of channel.
Perhaps limited to short lengths of channel.

CRUSHING RESISTANCE OF FRAGMENTED ICE COVERS

At first glance, the problem of ship passage through well fragmented ice might seem to be a typical problem of plastic indentation. One might think of the ice as a granular material with a Mohr-Coulomb type of failure criterion, and of the ship bow as a wedge or a cylindrical indenter that develops a slip line field in the horizontal plane. However, under typical conditions the ice cover is insufficiently thick for assumption of two-dimensional conditions, and in any case, there is a stress gradient through the thickness of the ice cover.

The first question concerns the yield criterion of the ice. It seems reasonable to assume a linear Mohr-Coulomb criterion, but very little is known about the c-ϕ properties of a floating mass of ice fragments.

Keinonen and Nyman (1978) made a few shear tests in a laboratory with model sea ice, and obtained a value of $\phi \simeq 47°$. They also reported a value of c = 11.3 N/m^2 (1.6x10^{-3} lbf/in.^{2}), although it is difficult to see from the data why c was not taken as zero. Angles of repose for piles of ice blocks were measured as 39° above water and 33° below water level. The value of ϕ = 47° is surprisingly high, suggesting that the blocks were well interlocked (confining pressures were up to about 1.4 kN/m^2, or 0.2 lbf/in.^{2}).

In other tests, the resistance to crushing has been measured by "bulldozing" floating ice horizontally with a vertical plate, confining the ice between the vertical walls of a parallel sided tank. Keinonen and Nyman (1978) obtained resistance values that corresponded to average pressures in the range 0.33 to 0.57 kN/m^2 (0.05 to 0.08 lbf/in.^{2}) with an ice cover 17.4 mm (6.85 in.) thick* and significant displacement. They calculated resistance using the standard soil mechanics formula for passive pressure on a retaining wall, but there are some questions about the way in which the formula was applied.

Tatinclaux et al. (1977) made a similar experiment, but they varied the pushing speed from 0.1 to 2.0 mm/s and found that the crushing resistance was inversely proportional to the pushing speed. The resistance was also apparently insensitive to the shape of the ice blocks. For ice covers that were 0.2 to 0.6 ft (61 to 183 mm) thick, the average crushing stress was *roughly* 1 lbf/in.^{2} (7 kN/m^2) at the lowest pushing speeds, and just less than 0.1 lbf/in.^{2} (0.7 kN/m^2) at the highest pushing speeds. The rate sensitivity of crushing strength was not explained, but a first guess is that it might represent a transition from static to dynamic friction within the ice mass (i.e. at high speed the ice mass near the plate would be fluidized). Another possibility is that crushing is localized near the plate only at high speeds where inertial effects appear.

In order to understand the full scale crushing process in floating ice, it is necessary to consider the mechanics of the process, but the necessary theory does not appear to have been developed. We therefore indulge in some preliminary theoretical speculation.

Consider a layer of uniformly fragmented ice floating in still water. If the thickness of a cohesionless layer t is significantly greater than the typical fragment size d, then there is necessarily lateral confinement of the layer (otherwise, it would simply spread out until it was one fragment thick). In the absence of an imposed stress system, the stresses in the ice cover are simply body forces induced by gravity. In the vertical direction, the normal stress σ_z in the layer above water level varies from zero at the upper surface to a maximum value at the water line, where

$$(\sigma_z)_{max} = \int_o^{t_1} \rho g \, dz \quad (1)$$

where ρ is the bulk density of the porous ice mass, t_1 is the freeboard of the ice layer, and σ_z is positive when compressive. If ρ and the porosity n do not vary with depth z, then

$$(\sigma_z)_{max} \doteq \rho g t_1 \quad (2)$$

also

$$\rho = \rho_i(1-n) \quad (3)$$

and therefore

$$(\sigma_z)_{max} = (1-n)\rho_i g t_1 \quad (4)$$

where ρ_i is the density of ice. Because ρ is assumed to be invariant with z, σ_z is proportional to depth z, i.e.

$$\sigma_z = (1-n)\rho_i g z. \quad (5)$$

Below the water line, σ_z varies from $(\sigma_z)_{max}$ at the waterline to zero at the base of the ice layer:

$$\sigma_z = (\sigma_z)_{max} - \rho' g(z-t_1) \quad (6)$$

*There is ambiguity in the original paper, where the term "ice thickness" refers to the thickness of individual blocks, and "ridge thickness" is the uniform thickness of the original brash ice layer.

where $\rho' g$ is the submerged unit weight of the ice. Thus

$$\sigma_z = (1-n)\rho_i g \left[t_1 - \left(\frac{\rho_w}{\rho_i} - 1\right)(z - t_1) \right] \quad (7)$$

where ρ_w is the water density.

The horizontal components of normal stress (σ_x and σ_y) for an elastic material that is confined laterally ($\epsilon_x = \epsilon_y = 0$) are:

$$\sigma_x = \sigma_y = \left(\frac{\nu}{1-\nu}\right)\sigma_z. \quad (8)$$

We do not know the value of ν, but from general physical knowledge we might guess a value around 0.3. Keinonen and Nyman (1978) deduced a value of 0.21 from their experiments. Alternatively, σ_x and σ_y can be expressed for a granular material in terms of horizontal pressure for Rankine limiting states of stress.

For the active state:

$$\sigma_x = \sigma_y = \left(\frac{1-\sin\phi}{1+\sin\phi}\right)\sigma_z. \quad (9)$$

For the passive state:

$$\sigma_x = \sigma_y = \left(\frac{1+\sin\phi}{1-\sin\phi}\right)\sigma_z. \quad (10)$$

A first guess might be $\phi \simeq 30°$, although Keinonen and Nyman (1978) found $\phi = 47°$, as already mentioned.

Taking reasonable values of ν and ϕ, the estimates of σ_x and σ_y given by eq 8 and 9 are in fair agreement.

Suppose now that the ice layer is pushed by a wide vertical plate until it fails. This is quite similar to the problem of passive pressure against a retaining wall, but there is a difference in that there are two free surfaces, and the variations of σ_x and σ_y with z are not monotonic. For a first approximation, we propose to consider separately the sections of plate that are above and below the waterline respectively, treating each in terms of a smooth retaining wall, and noting that there must be equality of σ_z at the junction between the two plate sections.

If the above-water section of plate is smooth and vertical, and if the ice behaves as a granular mass with $c = 0$, then according to classical Rankine theory the resistance per unit width developed by passive pressure, R_1, is

$$R_1 = \frac{1}{2}(1-n)\rho_i g t_1^2 \left(\frac{1+\sin\phi}{1-\sin\phi}\right). \quad (11)$$

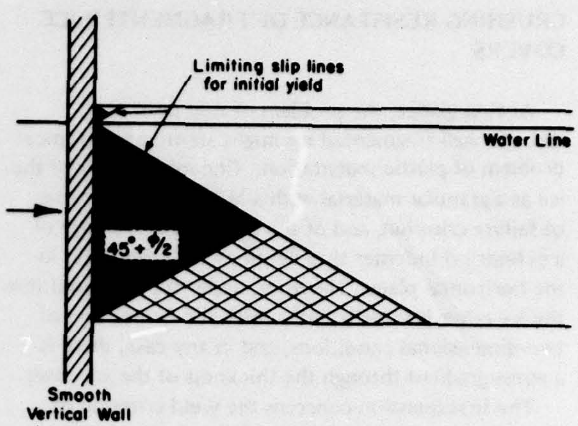


Figure 1. Schematic slip lines for a floating mass of cohesionless granular ice pushed horizontally by a wide, smooth, vertical wall.

Similarly, the resistance to the below-water section of plate, R_2, will be

$$R_2 = \frac{1}{2}(1-n)\rho_i g \left(\frac{\rho_w}{\rho_i} - 1\right) t_2^2 \left(\frac{1+\sin\phi}{1-\sin\phi}\right) \quad (12)$$

where t_2 is the depth of ice below water (the draft), and ϕ is assumed to have the same value in water and in air.

If the deviatoric failure strain of the ice above and below water is the same, the *displacement* required to initiate failure will be smaller for the above-water section of plate, which violates the implicit requirement that displacements should be equal at the waterline. However, once the failure has been initiated in accordance with a slip line field similar to that shown in Figure 1, then it seems reasonable to estimate the total resistance R per unit width as the sum of R_1 and R_2:

$$R = R_1 + R_2 = \frac{1}{2}(1-n)\rho_i g \left(\frac{1+\sin\phi}{1-\sin\phi}\right) t t_1$$

$$= \frac{1}{2}(1-n)\rho_i g \left(\frac{1+\sin\phi}{1-\sin\phi}\right)\left(1 - \frac{\rho_i}{\rho_w}\right) t^2 \quad (13)$$

since

$$\frac{t_1}{t_2} = \left(\frac{\rho_w}{\rho_i} - 1\right). \quad (14)$$

If R is expressed in terms of an average horizontal pressure R/t

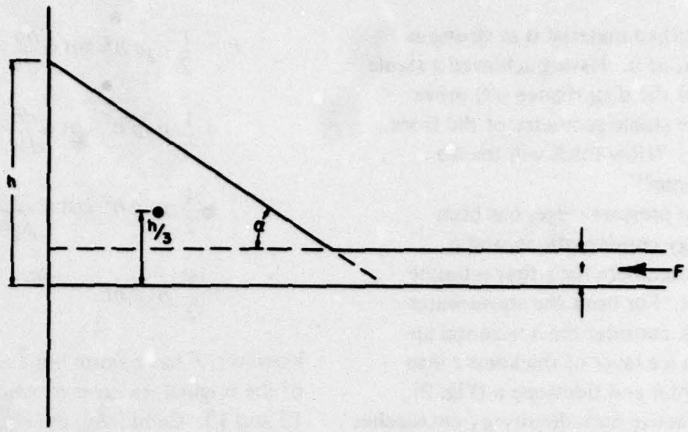


Figure 2. Idealized pressure ridge formed against a smooth vertical wall by horizontal thrusting across a frictionless base.

$$\frac{R}{t} = \frac{1}{2}(1-n)\rho_i g\left(\frac{1+\sin\phi}{1-\sin\phi}\right)\left(1-\frac{\rho_i}{\rho_w}\right)t. \qquad (15)$$

This illustrates that the average pressure of ice resistance will increase in proportion to the ice thickness.

At this stage it may be worthwhile to look at some numerical values.

A first guess at porosity might be around 40% or slightly higher, but apparently ice pressure ridges in the arctic typically have porosities less than 40%. The model ice block material used by Keinonen and Nyman had porosities in the range 32% to 37%. For present purposes, we can take $n = 0.35$, so that $(1-n)$ is 0.65.

For the unit weight of ice, $\rho_i g$, the value is 0.917 Mgf/m^3, or 8.99 kN/m^3 (57.2 lbf/ft^3).

The value of the coefficient of passive stress, $(1+\sin\phi)/(1-\sin\phi)$, varies appreciably with ϕ. Taking $\phi = 47°$, as found by Keinonen and Nyman (1978), the value of the function is 6.44. Taking $\phi = 30°$, which is a typical value for loose granular materials, the value of the function is 3.0.

For solid ice floating in fresh water, the value of $(\rho_w/\rho_i)-1$ is 0.0905. The value of $1-(\rho_i/\rho_w)$ is 0.083.

Using these values, calculations of resistance are in fairly good agreement with the test data of Keinonen and Nyman, but not with the data of Tatinclaux et al.

It might be noted that from eq 11, 12 and 14,

$$\frac{R_2}{R_1} = \frac{t_2}{t_1} = \left(\frac{\rho_w}{\rho_i} - 1\right)^{-1} \qquad (16)$$

Since $(\rho_w/\rho_i)-1 = 0.0905$ for fresh water, $R_2/R_1 \simeq 11$, so that R_1 contributes only 8% to the total force.

So far, the effects of cohesion between the ice blocks have been ignored. However, if there is finite uniform cohesion, the average resistance pressure R/t includes a term that is independent of the ice thickness t:

$$\frac{R}{t} = \frac{1}{2}(1-n)\rho_i g\left(\frac{1+\sin\phi}{1-\sin\phi}\right)\left(1-\frac{\rho_i}{\rho_w}\right)t + 2c\left(\frac{1+\sin\phi}{1-\sin\phi}\right)^{1/2} \qquad (17)$$

where c is the cohesion (intercept of a linear Mohr envelope) and ϕ is the effective value of internal friction for the cohesive condition of the ice.

When c becomes large, so that the second term of eq 17 dominates, the problem becomes similar to that of the unbroken ice sheet.

After initial yielding of the ice layer, the front of the disturbance tends to move away from the wall that is doing the pushing. With an infinitely wide wall and low pushing velocity, it is theoretically possible for yielding to be uniformly distributed with respect to normal distance from the wall, so that the ice would respond to pushing by thickening imperceptibly over its entire area. However, in more realistic situations where the distant ice is effectively restrained by inertia, water resistance, and "edge effects," it is more likely that yielding will be localized near the wall. Just how much the yielding is localized will probably depend on the residual strength of the disturbed material and on the pushing velocity.

For present purposes, assume that: 1) the pushing velocity is low enough for the problem to be considered in quasi-static terms, 2) the yielding initiates at the wall, and 3) the disturbed material is significantly weaker than the undisturbed material. Under these assumptions, it is to be expected that the ice will pile

up until the heap of disturbed material is as strong as the undisturbed ice ahead of it. Having achieved a stable cross section, the front of the disturbance will move out while maintaining the stable geometry of the front. The practical question is: "How thick will the ice ridge along the wall become?"

The piling of sea ice in pressure ridges has been treated by applying energy considerations, and a similar approach can be used here for a first estimate of maximum ridge height. For both the above-water and below-water sections, consider the horizontal encroachment of a uniform ice layer of thickness t into a triangular heap of height h and sideslope α (Fig. 2).

As the ice layer of effective bulk density ρ_1 encroaches horizontally a distance dx while pushing with a force F per unit width, it does work dW, where

$$dW = F dx \tag{18}$$

and it contributes to the pile an increment of mass dM per unit width, where

$$dM = \rho_1 t dx. \tag{19}$$

The total mass of the pile M is

$$M = \frac{1}{2} \rho_2 h^2 \cot\alpha \tag{20}$$

where ρ_2 is the bulk density of ice in the pile. Its potential energy E is

$$E = \frac{\rho_2 g}{6} h^3 \cot\alpha \tag{21}$$

and for a push by the ice layer through distance dx the potential energy increases by dE, where

$$dE = Mg dh = \frac{\rho_2 g}{2} h^2 \cot\alpha dh \tag{22}$$

If dissipative processes such as friction and fragmentation are ignored, then the work done by the ice layer must equal the energy gained by the pile:

$$dW = dE \tag{23}$$

i.e.

$$F dx = \frac{1}{2} \rho_2 g h^2 \cot\alpha dh \tag{24}$$

or

$$F = \frac{1}{2} \rho_2 g h^2 \cot\alpha \frac{dh}{dx}$$

$$= \frac{1}{2} \rho_2 g h^2 \cot\alpha \frac{dh}{dM} \cdot \frac{dM}{dx}$$

$$= \frac{1}{2} \rho_2 g h^2 \cot\alpha \frac{1}{\rho_2 h \cot\alpha} \cdot \rho_1 t$$

$$= \frac{1}{2} \rho_1 g h t. \tag{25}$$

However, F has a finite limit set by the crushing strength of the original ice layer R, where R is given by eq 11, 12 and 13. Combining the equilibrium conditions for the above-water and below-water portions, the maximum value of h/t is given by

$$R = \frac{1}{2} \rho_1 g h_{max} t \tag{26}$$

or

$$\left(\frac{h}{t}\right)_{max} = \frac{2R}{\rho_1 g t^2}. \tag{27}$$

Taking R from eq 13 and noting that, for the total thickness,

$$\rho_1 = (1-n) \rho_i \left(1 - \frac{\rho_i}{\rho_w}\right) \tag{28}$$

the maximum thickening is given by

$$\left(\frac{h}{t}\right)_{max} = \left(\frac{1+\sin\phi}{1-\sin\phi}\right) = K_p \tag{29}$$

where K_p is the coefficient of passive stress. With ϕ = 47°, K_p = 6.44 according to Rankine theory.

For another estimate of the maximum ridge thickness, consider a layer of brash ice which consists of two separate sections with different uniform thicknesses h_1 and h_2 (Fig. 3). The maximum force that can be developed by the thinner layer is given by a state of passive pressure:

$$(F_1)_{max} = \frac{1}{2} \rho_1 g h_1^2 K_{p1}. \tag{30}$$

The minimum force that can exist in the thicker layer is given by a state of active pressure:

$$(F_2)_{min} = \frac{1}{2} \rho_2 g h_2^2 K_{a2}. \tag{31}$$

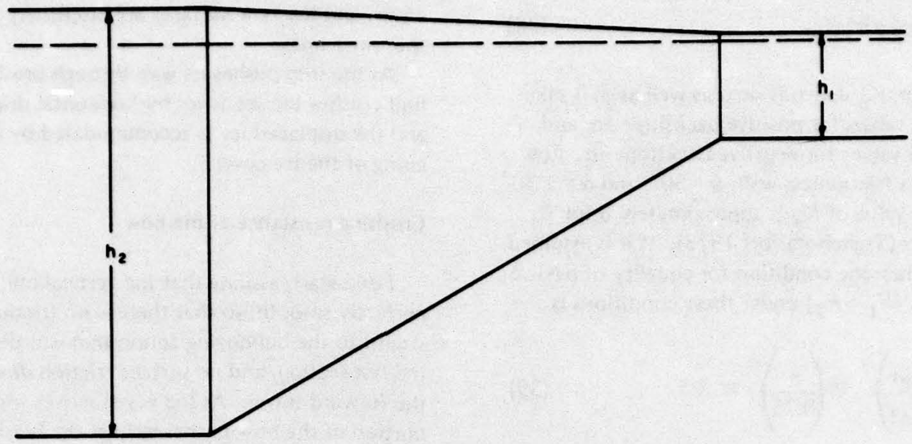


Figure 3. Transition between two ice layers of different thickness.

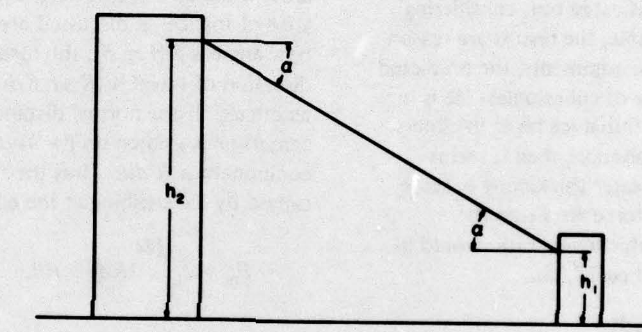


Figure 4. Retaining walls at the upper and lower ends of a uniform slope.

Thus, for equality of force in the two sections,

$$\frac{h_2}{h_1} \leqslant \left(\frac{\rho_1}{\rho_2}\frac{K_{p1}}{K_{a2}}\right)^{1/2}. \tag{32}$$

If it is assumed that $\rho_1 \simeq \rho_2$, then

$$\frac{h_2}{h_1} \leqslant \left(\frac{K_{p1}}{K_{a2}}\right)^{1/2}. \tag{33}$$

According to classical Rankine theory

$$K_p = \frac{1+\sin\phi}{1-\sin\phi} \tag{34}$$

$$K_a = \frac{1-\sin\phi}{1+\sin\phi}. \tag{35}$$

Therefore

$$\frac{h_2}{h_1} \leqslant \left[\left(\frac{1+\sin\phi_1}{1-\sin\phi_1}\right)\bigg/\left(\frac{1-\sin\phi_2}{1+\sin\phi_2}\right)\right]^{1/2}. \tag{36}$$

Taking $\phi_1 = 47°$ and $\phi_2 = 30°$, the maximum value of h_2/h_1 would be 4.4. If both sections have the same value of ϕ, then the maximum value of h_2/h_1 is K_p, as for the potential energy analysis (6.44 for $\phi = 47°$).

A lower limit estimate can be obtained by considering the transition slopes between the two ice layers, and applying the retaining wall analogy (Fig. 4). For equality of horizontal force at the two ends of the slopes, the *minimum* value of h_2/h_1 occurs when there is passive pressure at both ends (assuming that pressure must be passive at the lower end). The forces at the two walls are

$$F_1 = \frac{1}{2}\rho_1 g h_1^2 K_{p1} \tag{37}$$

and

$$F_2 = \frac{1}{2}\rho_2 g h_2^2 K_{p2}. \qquad (38)$$

In this situation, K_p depends on α as well as ϕ; it takes relatively large values for positive backslope $+\alpha$, and relatively small values for negative backslope $-\alpha$. For example, with a frictionless wall, $\phi = 30°$, and $\alpha = \pm 30°$, the theoretical value of K_p is approximately 8 for $+\alpha$ and 0.75 for $-\alpha$ (Tschebotarioff 1973). If it is assumed that $\rho_1 = \rho_2$, then the condition for equality of passive pressure forces ($F_1 = F_2$) under these conditions is

$$\frac{h_2}{h_1} = \left(\frac{K_{p1}}{K_{p2}}\right)^{1/2} \cong \left(\frac{8}{0.75}\right)^{1/2} \cong 3.3 \,. \qquad (39)$$

If Tschebotarioff's tabulated K_p values for a curved failure surface are substituted for the values used above, the value of h_2/h_1 is 4.3.

These analyses are unsophisticated but, considering how little information is available, the results are reasonably consistent. From separate arguments, the predicted thickening in the pressure zone of cohesionless ice is in the range 3.3 to 6.4 times the initial ice layer thickness.

If the initial ice layer has cohesion, then it seems possible that there could be greater thickening because of the increase in the pushing force R. From the potential energy analysis, the thickening ratio would be given by combining eq 27 with eq 17, i.e.

$$\left(\frac{h}{t}\right)_{max} = K_p + \frac{4cK_p^{1/2}}{(1-n)\left(1-\frac{\rho_i}{\rho_w}\right)\rho_i g t} \,. \qquad (40)$$

From the balancing of active and passive pressure in layers of thickness h_1 and h_2, a relatively simple result can be obtained by making the reasonable assumption that initial cohesion is negligible in the thickened layer for the active stress state. If it is further assumed that $\rho_2 \simeq \rho_1$, then

$$\frac{h_2}{h_1} \cong \left[\frac{K_{p1}}{K_{a2}} + \frac{4c}{(1-n)\left(1-\frac{\rho_i}{\rho_w}\right)\rho_i g t} \cdot \frac{K_{p1}^{1/2}}{K_{a2}}\right]^{1/2} . \qquad (41)$$

However, without knowing something about the magnitude of c, it is not profitable to pursue the matter at this stage.

RESISTANCE TO SHIP PASSAGE BY BROKEN ICE

The linear dimensions of a Great Lakes bulk carrier are large compared to the thickness of a typical ice cover, and the bow surfaces are essentially vertical near the water line.

As the ship pushes its way through brash ice, the hull crushes the ice cover by horizontal displacement, and the displaced ice is accommodated by local thickening of the ice cover.

Crushing resistance at the bow

For a start, assume that the vertical hull surface is perfectly smooth, so that there is no frictional restraint to the bulldozing action that was discussed in the last section, and no surface friction *directly* resisting the forward travel. As the vessel moves forward, each portion of the bow pushes against the brash ice and exerts a normal force of R per unit length on the ice (Fig. 5). If the hull surface is perfectly smooth, there is no tangential stress. The force R is the same all around the bow, being determined by the yield resistance of the ice, as discussed previously. If the local bow angle is β (Fig. 5), the force component in the direction of travel is ($R \sin\beta\, ds$) for each element of length ds. If the normal distance from the ship's centerline is y, then $\sin\beta = dy/ds$ and the forward force component is $R\,dy$. Thus the total resistance to motion caused by ice crushing at the bow is

$$F_{bc} = 2\int_0^{B/2} R\,dy = RB. \qquad (42)$$

For cohesionless ice

$$F_{bc} = \frac{B}{2}(1-n)\rho_i g\left(1-\frac{\rho_i}{\rho_w}\right)K_p t^2 \qquad (43)$$

and for ice with cohesion c

$$F_{bc} = \frac{B}{2}(1-n)\rho_i g\left(1-\frac{\rho_i}{\rho_w}\right)K_p t^2 + 2BtcK_p^{1/2}. \qquad (44)$$

Tangential friction at the bow

Interface friction and crushing resistance are interrelated processes, but for present purposes it is simpler to treat them separately. Here we first assume that horizontal sliding between the ice and the ship's bow develops a tangential frictional force per unit length P_{bf} that is

$$P_{bf} = \mu_e R \qquad (45)$$

where μ_e is an effective friction coefficient and R is the normal force per unit length, as determined previously in the analysis for ice crushing (Fig. 6). The forward

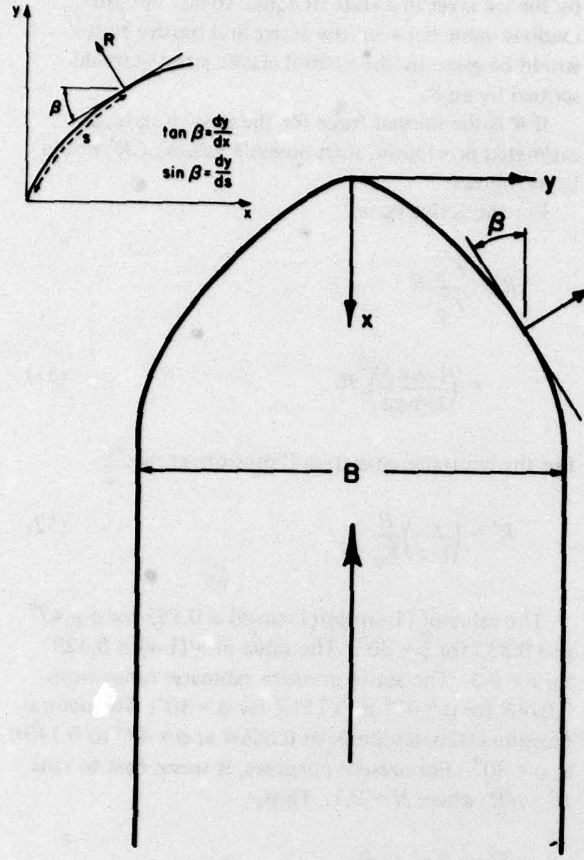


Figure 5. Plan diagram of a ship bow thrusting through brash ice.

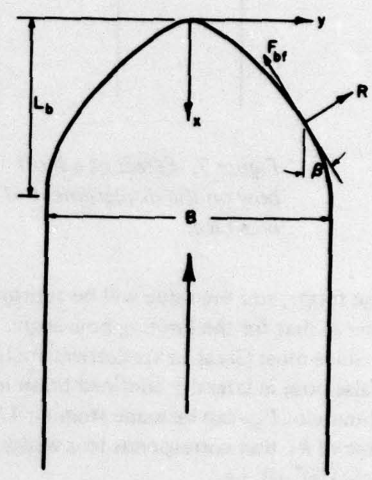


Figure 6. Normal and tangential forces acting on the bow section of a ship hull.

component of P_{bf} is P_{bf} cosβ and, since cosβ = dx/ds, the total resistance to motion that is developed by bow friction is

$$F_{bf} = 2\int_0^{L_1} \mu_e R dx = 2\mu_e R L_1 \quad (46)$$

where L_1 is the centerline length of the bow section. For comparison of the contributions made by crushing resistance F_{bc} and friction resistance F_{bf}, eq 46 can be rewritten as

$$F_{bf} = 2\mu_e R k_1 B \quad (47)$$

where

$$k_1 = L_1/B \quad (48)$$

Equation 47 implies that bow friction decreases as the bow angle increases, which agrees qualitatively with the predictions of plasticity theory for wedge indentation (there are differences of detail, however). For a conventional ship, k_1 might be 3 or more, but for a Great Lakes carrier it could be less than unity. This brings in another consideration.

If $k_1 \geq 1$ and the hull is fairly smooth, then it is quite likely that ice will slide past the bow by slip at the interface, as assumed above. By contrast, if the bow is very stubby and of finite roughness, it is likely that a "false nose" of ice will form in front of the vessel, and the limiting slip line will be within the ice mass (Fig. 7). In a thick ice layer, where shear displacements take place by slip along vertical planes, the expected angle at the tip of an idealized false nose is (45°−ϕ/2) for the half angle, or (90°−ϕ) for the total included angle (this expectation derives from established plasticity theory for two-dimensional indentation of a half-space). By analogy with two-dimensional wedge indentation according to plasticity theory, it is expected that bow friction will not change once a false

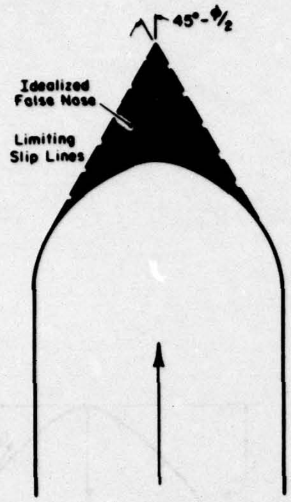


Figure 7. Effect of a bluff bow on the displacement of brash ice.

nose forms, and the value will be approximately the same as that for the limiting bow angle.

Since most Great Lakes carriers are likely to form a false nose in laterally confined brash ice, a preliminary estimate of F_{bf} can be made from eq 47 by taking a value of k_1 that corresponds to a wedge bow of included angle (90°–ϕ), i.e.

$$k_1 = \cot(45° - \phi/2). \tag{49}$$

If $\phi = 47°$, $k_1 = 1.27$. For $\phi = 30°$, $k_1 = 0.866$. Taking $\mu_e = 0.2$, estimated values for F_{bf} are 0.5 RB for $\phi = 47°$ and 0.35 RB for $\phi = 30°$.

Hull friction aft of the bow section

If the widest part of the hull begins immediately aft of the bow section, there should be no more crushing resistance aft of the bow. The frictional resistance along this part of the hull, F_{hf}, can be estimated as

$$F_{hf} = 2\mu_e L_2 R' \tag{50}$$

where L_2 is the length of the hull which is aft of the bow section and in continuous contact with the ice (the length of the parallel sides for a laker). R' is the normal force per unit length of the hull that is exerted by the ice.

R' is expected to be less than the crushing force R, which represents passive pressure. The minimum value of R' in confined ice is likely to be the force developed by the ice layer in a state of active stress. An intermediate value between the active and passive states would be given by the neutral elastic stress state described by eq 8.

If R is the normal force for the passive state, as estimated previously, then possible values of R' would be as follows.

For the active state:

$$R' = \frac{K_a}{K_p} R$$

$$= \left(\frac{1-\sin\phi}{1+\sin\phi}\right)^2 R. \tag{51}$$

For the neutral elastic state ("pressure at rest"):

$$R' = \left(\frac{\nu}{1-\nu}\right)\frac{R}{K_p}. \tag{52}$$

The value of $(1-\sin\phi)/(1+\sin\phi)$ is 0.155 for $\phi = 47°$ and 0.333 for $\phi = 30°$. The value of $\nu/(1-\nu)$ is 0.429 for $\nu = 0.3$. The active pressure estimates range from $0.024R$ for $\phi = 47°$ to $0.111R$ for $\phi = 30°$. The neutral pressure estimates are from $0.066R$ at $\phi = 47°$ to $0.143R$ at $\phi = 30°$. For present purposes, it seems best to take $R' = NR$, where $N \sim 0.1$. Thus,

$$F_{hf} = 2\mu_e L_2 R'$$

$$= 2\mu_e k_2 BNR \tag{53}$$

where

$$k_2 = L_2/B. \tag{54}$$

For a Great Lakes carrier, a typical value of k_2 might be about 8 or 9. With $\mu_e = 0.2$ and $N = 0.1$, the estimated value of F_{hf} is $0.32RB$ to $0.36RB$, or roughly comparable to F_{bf}.

Total ship resistance from brash ice

The total ship resistance from brash ice, F_T, is the sum of the crushing resistance at the bow, the bow friction, and friction along the main hull. The estimates are made on the assumption that the ship speed is low, and with this assumption F_T ought to approximate the total resistance of the ship (ignoring ordinary hydrodynamic resistance). Expressing F_T in terms of R:

$$F_T = F_{bc} + F_{bf} + F_{hf}$$

$$= RB+2\mu_e k_1 RB+2\mu_e k_2 NRB$$

$$= RB[1+2\mu_e(k_1+k_2 N)]. \qquad (55)$$

In this equation, R is the force per unit length given by eq 15 or eq 17. The friction coefficient μ_e might be approximately 0.2. The effective value of k_1 is 1/2 cot $(45°-\phi/2)$, or $k_1 \simeq 1$. For a typical Great Lakes carrier, $k_2 N \sim 1$. Thus for a rough estimate it might be sufficient to take

$$F_T \simeq 1.8\, RB. \qquad (56)$$

COMPARISON OF SHIP THRUST AND ICE RESISTANCE

When a vessel is pushing slowly through brash ice at constant speed, its thrust equals the ice resistance. In the limiting condition, when the vessel is using all its available thrust

$$F_T = k_3 P_s \qquad (57)$$

where P_s is the maximum shaft horsepower and k_3 is the thrust per unit power.

Dividing through eq 57 by B

$$R[1+2\mu_e(k_1+k_2 N)] = k_3(P_s/B). \qquad (58)$$

Figure 8 is a plot of shaft horsepower P_s against beam B for Great Lakes bulk freighters. While P_s is clearly not proportional to B, representative values of (P_s/B) can be taken from the plot. The ships with the lowest power per unit width have 30 hp/ft, while the biggest and most powerful ships have about 150 hp/ft.

The coefficient k_3 depends on a number of factors, but for present purposes we can take 20 lbf/shp as a reasonable (but maybe optimistic) value for estimating purposes. Combining this value of k_3 with the approximation of (F_T/B) from eq 56:

$$1.8\, R \simeq 20\, (P_s/B) \qquad (59)$$

where (P_s/B) is in hp/ft and R is in lbf/ft. From eq 59, estimated limit values of R would be approximately 1700 lbf/ft for a ship with 150 hp/ft, 1100 lbf/ft for 100 hp/ft, and 330 lbf/ft for 30 hp/ft. These figures can be compared with resistance forces calculated from ice properties.

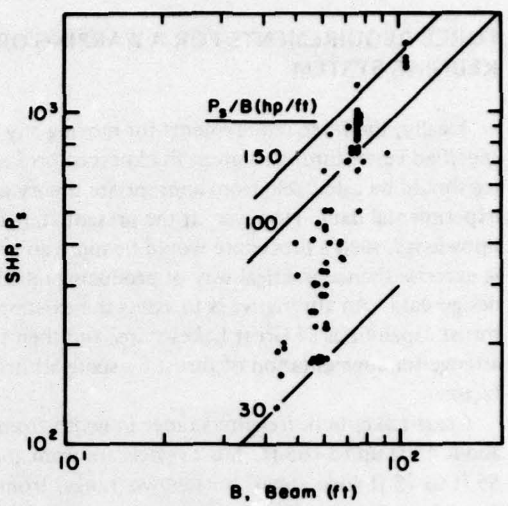


Figure 8. Plot of shaft power against beam for Great Lakes bulk freighters.

If the ice is cohesionless, R can be estimated from eq 13 as

$$R = \frac{1}{2}(1-n)\rho_i g \left(\frac{1+\sin\phi}{1-\sin\phi}\right)\left(1-\rho_i/\rho_w\right) t^2. \qquad (60)$$

Taking $n = 0.35$ and substituting for ρ_i and ρ_w

$$R = 1.53\left(\frac{1+\sin\phi}{1-\sin\phi}\right) t^2 \quad \text{lbf/ft} \qquad (61)$$

when t is in feet. If $\phi = 47°$ and there is a 6-ft layer of brash ice, the calculated value of R is 360 lbf/ft. According to the calculations above, this should be enough to immobilize the low powered vessels, but not the big, powerful ships. To stop the biggest and most powerful vessels in completely cohesionless brash ice, it would take 13 ft according to theory (and assuming the ships really do develop 20 lbf/shp). If the ice has some cohesion, then the resistance R becomes correspondingly higher (see eq 17). For example, with a 6-ft layer of ice the resistance would be doubled with $c = 12$ lbf/ft² (0.08 lbf/in.² or 0.57 kN/m²).

To conclude, it seems that the theory developed here for cohesionless brash ice may be quite realistic. However, for practical application the big difficulty is that there is likely to be cohesion, which can vary unpredictably within wide limits. The whole matter needs some research.

FORCE REQUIREMENTS FOR A WARPING OR KEDGING SYSTEM

Ideally, the force requirements for moving any specified vessel through a given thickness of broken ice should be calculable from appropriate theory and experimental data. However, at the present state of knowledge, such a procedure would be more an academic exercise than a practical way of producing reliable design data. An alternative is to assess the existing thrust capabilities of Great Lakes ships, and then to arrange for augmentation of thrust by some arbitrary factor.

Great Lakes bulk freighters range in width from about 43 ft up to 105 ft. Most vessels are from about 55 ft to 75 ft wide. Shaft horsepower ranges from less than 2000 hp to 16,000 hp. The low-speed thrust, or bollard pull, of a vessel depends on the shaft power, the propeller design, and certain operating characteristics, but for present purposes it suffices to assume that there is a fixed ratio between thrust and shaft power. The ratio is perhaps about 15 to 20 lbf/shp under favorable conditions (some tugs with special shrouded propellers are supposed to develop more than 30 lbf/shp). As was mentioned earlier (see Fig. 8), the power per unit width for Great Lakes carriers ranges from 30 shp/ft to 150 shp/ft, and using a ratio of 20 lbf/shp this converts to thrust per unit width in the range 600 to 3000 lbf/ft. As a matter of interest, the most powerful freighters have slightly more power per unit width than a large Great Lakes icebreaker.

When a vessel is just on the brink of immobilization, the resistance is equal to the maximum thrust T. For low-powered vessels T is less than 40,000 lbf, or 20 tons. For the most powerful vessels, T is about 300,000 lbf, or 150 tons.

In order to decide the force capability of a towing system F_p, it is reasonable to think in terms of thrust augmentation, such that

$$F_p = k_4 T$$

where k_4 is a dimensionless factor. When a vessel moves through an ice-clogged channel under its own screw power with assistance from a towing system, the total propulsive force F_p' is

$$F_p' = (1+k_4)T.$$

In deciding what values to accept for k_4, common sense defines the broad limits. For example, with $k_4 = 0.1$ the towing system would not make a significant difference, whereas with $k_4 = 10$ it is quite possible that the hull could be damaged in heavy ice, or the deck fittings might be ripped out. With $k_4 = 1.0$ the effect would be the same as a doubling of the vessel's shaft power. With $k_4 = 2.0$ the effect would be equivalent to having three times the original shaft power, and so on. A reasonable range for k_4 might be 0.5 to 2.0.

A complicating factor is that T varies widely over the whole range of ships, but a towing system has to serve all the ships using the waterway. For some complete systems, and for certain parts of other systems, the force capabilities have to match the needs of the largest vessels. Examples of things that have to be designed to the maximum force limit include permanent anchors, warping tugs, fixed winches and their cables. There are other components that need only match the force requirements of particular vessels, e.g. on-board winches and capstans, and on-board winch cables. Clearly the first priority is to establish reasonable force levels for those components that have to serve all types of vessels, up to the largest ones using the system.

The largest value of T for Great Lakes freighters (T_{max}) is perhaps somewhere in the range 120 to 160 tons force, depending on the exact value of the thrust/power ratio. For estimating purposes, we take here a value of $T_{max} = 150$ tons.

In order to decide what might be a realistic short-range goal for the pulling force of a tow system F_p, we can consider the maximum towing force that is currently available from a large icebreaker. With 10,000 shp, the total low speed thrust of an icebreaker might be about 100 tons force. According to the foregoing analysis, an icebreaker does not enjoy much of an advantage over a freighter in brash ice—the only major difference is that it does not have much afterbody resistance. Using the figures that were suggested earlier, this means that for equal beam the resistance of an icebreaker is only lower than that of a freighter by a factor of (1.4/1.8), or 0.78. If the conditions are just on the brink of immobilizing a big freighter with the same power/width ratio as the icebreaker (140 shp/ft), then the breaker only has a surplus thrust equal to 22% of its total thrust, say 22 tons. This is enough to provide a maneuvering advantage over a powerful freighter, but not enough to provide a useful pull by direct towing. However, the icebreaker could move ahead of the freighter and tow with its winch. With existing equipment, the icebreaker's winch and cable is rated for a maximum working load of 50 tons.

To sum up these icebreaker considerations, the maximum winch capability is 50 tons, and the direct towing capability might be about the same when conditions are such that *typical* freighters are just immobilized.

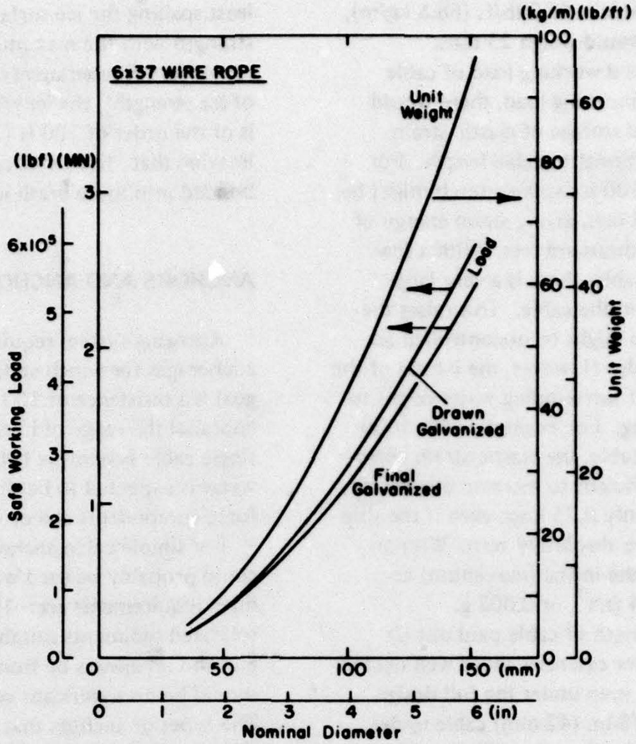


Figure 9. Safe working load and unit weight as functions of diameter for 6×37 galvanized wire rope.

If F_p were to be taken as 50 tons, the value of k_4 relative to T_{max} would be only 0.33. However, relative to T for freighters with about 9000 shp it would be over 0.5, and for the 2000 shp vessels it would be 2.5.

Weighing all of these considerations, and anticipating the difficulties of providing very large towing forces, it is suggested that a reasonable short-term goal for F_p would be 100 tons force. For the most powerful freighters, this would increase the thrust by a factor of 1.67. For the lowest powered ships the potential thrust increase would be by a factor of more than 6.

TOW CABLE REQUIREMENTS

Using an operational scheme in which each vessel provides its own tow cable, the strength of the cable would be matched to the towing requirements of the vessel carrying the cable. For an operational scheme where a single towing cable has to serve all vessels using the system, the strength has to match the maximum design force of the system.

The value accepted here as a design force for short term development is 100 tons. However, it seems desirable to consider a force range that spans this value, say from 50 to 200 tons. For this force range, use of synthetic fibre ropes is probably ruled out, and attention concentrates on stranded steel cables from about 2.5 to 5 in. in diameter.

The main requirement here is to estimate the diameter and the weight of tow cable. It is not necessary at this stage to consider the fine details of cable design and their relevance to particular operations. It will simply be assumed that the cable needs the flexibility, fatigue resistance, and wear resistance appropriate to general towing and winching applications. Data have been taken for 6x 37 classification galvanized cable (API Spec. 9A), and a safety factor of 5 has been adopted for determination of safe working load.

Figure 9 gives safe working load and unit weight of 6x 37 wire rope as functions of diameter. For the design working load of 100 tons, a cable of 3-1/2-in. (90-mm) diameter would serve. The unit weight for cable of this size is 22.7 lb/ft (33.8 kg/m), or 11.35 tons per thousand feet.

For a 50-ton system, or for a 100-ton system that uses a doubled cable, the required diameter is 1 5/8 in. (42 mm). The unit weight for cable of this size is 4.88 lb/ft (7.26 kg/m).

For a high force system with a 200-ton capacity, the required cable diameter would be 5 in. (128 mm).

The unit weight of this cable is 46.2 lb/ft (68.8 kg/m), so that 1000 ft of cable would weigh 23 tons.

Even though the planned working load of cable would be well below the breaking load, there would be appreciable stretch and storage of elastic strain energy, with both proportional to cable length. For a 3.5-in. cable loaded to 100 tons, the stretch might be about 2.5 ft per thousand feet, giving strain energy of about 500,000 ft-lbf per thousand feet. With a few thousand feet of loaded cable, there is a very large amount of energy stored in the cable. This raises the question of whether there might be uncontrolled accelerations as the ice yields. However, the inertia of the ship and its added mass of surrounding water ought to provide sufficient damping. For example, with 5000 ft of fully loaded 3.5-in. cable, the elastic strain energy in the cable would be sufficient to increase the velocity of a 50,000-ton ship by only 0.75 knot even if the ship resistance is assumed to be absolutely zero. With an initial force of 100 tons, the initial (maximum) acceleration would be 0.064 ft/s^2, or 0.002 g.

With any significant length of cable paid out (> 1000 ft), the sag of the free catenary could well exceed the depth of the channel, even under the full design load. For example, a 1-5/8-in. (42-mm) cable under a 50-ton pull would sag 24.4 ft with 1000 ft paid out, 97.8 ft with 2000 ft out, 221 ft with 3000 ft out. With 3-1/2-in. (90-mm) cable under 100-ton load, the corresponding sag values are greater. With thick, continuous ice, the cable might be supported on the ice surface. On the other hand, a cable dragging across cohesionless brash ice might dig its way through and sink to the bed of the channel. If the cable does sink, then it is likely to rest partly on the bed of the channel, even during the active tow.

Another question is whether or not the cable would freeze into the ice to the extent that it could not easily be freed. If the cable were to be left on top of the ice, flooded, and then frozen along its length, then it certainly could develop a total bond force far in excess of the tensile strength of the cable. However, even if this were to happen the cable could be peeled free by pulling at a sharp angle from one or both ends. A more awkward situation might arise if the cable were allowed to pass through the ice cover at a shallow angle and freeze in, although this seems an unlikely thing in any place where ship traffic is contemplated. If a cable were to be frozen in, and if its surface was untreated to the extent that a perfect bond could form, the breakout conditions would be governed by the strength of the ice itself. For a direct pull, the working load on a typical cable gives longitudinal strain several times greater than the tensile fracture strain of ice, and on this basis alone there is some hope for freeing the cable, or at least spalling the ice surface. Comparing the bond strength with the maximum safe working load in the cable (for representative cable sizes and credible values of ice strength), the length of cable that could be freed is of the order of 100 ft (30 m). Actually, it is hard to imagine that "live" tow cables would become securely bonded into loose brash ice.

ANCHORS AND ANCHORAGES

A towing system requires either cable anchors, or anchorages for winch stations. The immediate design goal is a resistance of 100 tons, but for more general appraisal the range of horizontal working forces for single cable systems is 100,000 to 400,000 lbf. The water is expected to be shallow, and significant vertical force components can probably be avoided.

For simple cable anchors, conventional burial anchors could probably be used without much trouble. The main requirements are: 1) the bed should have unconsolidated sediments suitable for embedment, 2) the pull should always be from the same direction, 3) there should be no significant vertical component of force. The types of anchors that could be considered include the hinged-fluke type, the various scoop or pick types, and mushroom mooring anchors. Anchor designs that might be of interest include stockless, Moorfast, Danforth, Stato, Boss, Stevin, Hook, Digger, Bruce, and Admiralty Mooring. If an efficient anchor is properly embedded and set, it is probably safe to assume that the holding force will be equal to at least 20 times the weight of the anchor. Since available anchors are up to 60 tons (55 tonnes) in weight, it is possible to develop holding forces exceeding 1000 tons with burial anchors. Provided that the bed material is suitable, it ought to be relatively easy to obtain deep embedment by jetting or dredging an anchor pocket.

Another possibility for a permanent anchor is a cluster of piles driven into the bed. The merits of such a method would have to be decided by engineering analysis for particular site conditions, but in studies of deep ocean mooring systems, Valent et al. (1976) decided that piles would be undesirable as anchors in soft sediments. In a hard or rocky bed, drilled piles or large rockbolts might be very suitable.

Mass concrete gravity anchors would also be capable of developing the required levels of resistance. A sufficiently large anchor of the simple deadweight type would suffice, but it would probably be better to set a block into some kind of bed excavation, or alternatively to settle it into the bed sediment by jetting or suction dredging. As a rough rule of thumb, the horizontal holding capacity of a concrete block is often considered

to be about 50% of the weight of the block, although Valent et al. (1976) took more conservative ratios in their study of large moorings. A buried block ought to be more efficient, but a soil mechanics analysis for local conditions is needed for quantitative assessment.

The anchorage problem is more difficult where winches or pulleys have to be set on, or beneath, the bed of the channel. In these cases it is not only necessary to develop the required resistance; it is also necessary to ensure that the foundations and housings will not be displaced or rotated. One possibility would be to precast a concrete gravity structure, float or barge it to the site, sink it into a prepared excavation, and lock it into place with engineered backfill. Another possibility would be to drive sheet piling and build a caisson. A design analysis for bed structures is beyond the scope of this study, but general feasibility has to be considered.

The first question concerns resistance to horizontal force in the range 100,000 to 400,000 lbf. With a gravity structure, the required reaction would probably have to be provided by shear between the foundation material and the structure, and by thrust of the end wall and vertical surfaces against the adjacent soil mass. For a very rough estimate of base friction, an effective value between 0.3 and 0.5 might be assumed for the ratio of horizontal resistance to weight of concrete in air. For a rough first estimate of the thrust reaction, a pressure of 1000 lbf/ft^2 might be assumed; this is approximately the passive pressure for a "long" wall in a submerged soil having $c \approx 400\ lbf/ft^2$; $\phi \approx 6°$, and submerged unit weight $\approx 60\ lbf/ft^3$ (closer estimates can be made from standard equations for retaining walls). Using these values for a simple box structure buried directly in bed sediments, a 200-ton horizontal resistance could be developed by a 200-ton concrete structure with end wall area between 200 and 300 ft^2. The problem of obtaining horizontal resistance of 400,000 lbf is not trivial, nor is it insoluble. A careful design, perhaps utilizing rockfill, is called for.

The second question concerns bearing capacity of the structural foundation, and avoidance of differential settlement. It can probably be assumed that adequate bearing capacity is attainable with either a spread footing or a pile foundation, provided that there are no severe tilting movements on the structure. The latter consideration is a serious one, since inappropriate design could easily create a large tilting moment. However, if the towing cable is led down to a low point of the structure before load is transferred, then large moments can be avoided. For example, with a horizontal-axis winch winding to the bottom of the drum, cable could be led into the box structure through an inclined pipe. The pipe itself would have to be well founded, perhaps in rock fill, and it might be fitted with a system of rollers and scrubbers for the cable.

FORCE AND POWER REQUIREMENTS FOR WINCHES AND WINDLASSES

The primary concern in the selection of winches is the force and torque capability. The towing force suggested earlier is 100 tons, but it is of interest to keep in mind a wider force range, from 50 to 200 tons.

There are two types of machines to be considered: 1) winches, which typically wind cable onto one or more drums, 2) windlasses, which wind cable or chain and feed it to a storage reel or a storage locker.

The required force capability of 50 to 200 tons is beyond the typical range for deck machinery on ordinary commercial vessels. However, it corresponds quite closely to the force range for existing marine traction winches and windlasses, as used in towing, anchor handling, lay barge anchoring, and heavy mooring systems. It is also comparable to the force levels developed by winches used for laying underwater pipelines by the bottom-pull method. The maximum stall force of commercially available winches and windlasses approaches 300 tons on the bare drum, with static braking forces up to 500 tons or so. However, the recommended working load, or dynamic load, may be only about half the stall force, and there are further significant reductions when the drum is full, or partly full, with spooled cable. The requirement of 100 tons dynamic pull can certainly be met by existing equipment. The upper limit of the force range considered here, 200 tons, is close to the limit of existing equipment, but there are both winches and windlasses capable of providing dynamic pull at this force level.

The power requirements for the winch depend on the towing speed. Figure 10 gives power requirement as a function of towing speed for different force levels, and for two levels of drive system efficiency. The winch itself has high efficiency, but with hydraulic or diesel-electric drive the efficiency relative to the primary power source might be around 65%. Typical marine winches and windlasses are of fairly low pow[illegible] few hundred horsepower), and consequently their [illegible]imum speeds at full load are quite low - less than 1 knot. With a 100-ton force and a 1-knot (101-ft/min) towing speed, the power requirement would be just over 600 hp at 100% efficiency, and about 940 hp at 65% efficiency. This is below the power and speed limits already reached in anchoring winches for lay barges. Provision of high power and high cable speed is not expected to be a great problem, since oilfield drawworks are rated at up to 3000 hp.

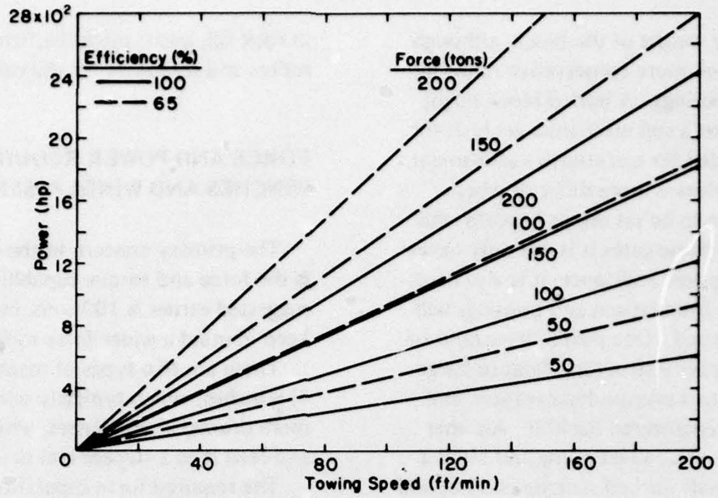


Figure 10. Power requirements of winches as a function of towing speed, for various force levels and two levels of transmission efficiency.

One of the required design features for the winch is ability to handle big lengths of large diameter cable, perhaps 5000 ft or more of 3.5-in. cable (twice the length and about half the diameter for a double-cable system). This capability exists on high force winches and windlasses; cable may be fed from the main winch drum to a secondary storage drum, or it may be passed to a storage locker. Some existing large winches can spool up to 10,000 ft of 3-in. cable.

Another required feature in a universal system would be tension control, since it would be dangerous to have the full force capability transmitted to some of the smaller vessels that might use the system. This again is a feature that already has been developed, both for maintaining tension in quasi-static situations and for limiting dynamic forces in rough-water towing.

PICKUP OR TRANSFER OF CABLES

For a system which requires a passing ship to attach itself to a cable that is already in place, an obvious arrangement is to have a buoyed leader cable. The free end of the leader cable, or branch cable, is attached to a buoy, which itself is moored either by the cable or by a separate anchor, depending on the details of the system. On arrival, a transient vessel picks up the buoy and the cable and makes the attachment. At the end of the tow, it drops the buoy at the designated spot. Perhaps the main problem here is the ability of the buoy to withstand ice forces and ice drift, and to avoid becoming frozen into the ice. There is some experience of designing buoys for use in ice, and it is not expected that the buoy problem would be an insuperable one.

As a last resort, the buoy could be parked on the channel bed and brought to the surface on radio command, using an ice-penetrating proboscis to break through the ice cover.

One of the suggested systems (F) envisages use of a capstan on board the passing vessel. The fixed cable has to be picked up at some point between its two end anchors, and then engaged to the capstan. The traditional single-drum capstan might be very difficult to use under these circumstances. An alternative would be to lay the cable in between two horizontal lines of drive pulleys, and then to force the two sets of pulleys together by a hydraulic mechanism so that the cable follows a sinuous path.

In some systems the tow cable would be passed directly by a local tug. If the tug is able to maneuver under the bow of the transient vessel, this is a straightforward operation. If close-quarter maneuvering became a problem in heavy ice, it might be necessary to employ a line-throwing device of some kind. In some locations it might be possible to pass the tow line before entering the channel where ice is tightly jammed.

There are also exotic possibilities for passing light leader lines by helicopter or hovercraft, but these do not seem very practical under foreseeable circumstances.

GENERAL APPRAISAL

Virtually all of the concepts that were outlined at the beginning of this report could be put into practical operation. However, the development effort, the expense, and the ease of operation probably vary widely among the different schemes.

Perhaps the most appealing concept for a full scale *experiment* is (A), the warping tug system, since it calls for rather little capital expenditure and virtually no preliminary development effort. The required tug could probably be chartered, although it might be necessary to come to a special arrangement for installation of a high-force traction winch. The winch itself might have to be purchased or leased for the test project. If the tow were to be used over a single, relatively short, section of channel, then only two anchors would be required. Since these would have to hold against a force from only one direction, they could be conventional anchors of about 5 ton size. If, however, the tug were required to transit a ship in stages by stepping from one anchor to another, then the intermediate anchors would have to take pull from two opposite directions, depending on which way the traffic was moving. Well set blocks or mushrooms might be used for this job. Whatever anchors might be used, it would be worthwhile to set them deeply by jetting or by suction.

For *routine operation* over the longer term, the chain ferry barge (C) seems attractive. The investment is modest, and operating costs ought to be reasonable. The basic requirements are: 1) a set of anchors, 2) a chain lying on the channel bed, 3) a special hull fitted with a primary power plant and a large windlass. The main thing about the hull is that it should be properly stressed and balanced to take the towing forces from the chain and the ship's hawser.

All the schemes that depend on underwater machinery seem dubious. The required technology and equipment is all readily available, but costs would be high and there would probably be many maintenance problems. The various schemes that call for transient vessels to tow themselves using their own on-board machinery are also questionable; sooner or later some vessel or crew will foul up the system.

Concept (J), the chain ferry plow, is rather interesting, even though it really relates to the problem of *ice clearing*, which was the subject of an earlier study (Mellor et al. 1978). The very high efficiency of a winch or windlass means that ice could be cleared by plowing for very modest expenditures of energy and power, especially if plowing operations could keep pace with ice accumulation.

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